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# PERFORMANCE INVESTIGATION WITH OIL-INJECTION TO COMPRESSION CHAMBERS ON CO<sub>2</sub>-SCROLL COMPRESSOR

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## ABSTRACT

Recently, from the viewpoint of global warming, natural gas CO<sub>2</sub> is considered as a main refrigerant for air conditioning. CO<sub>2</sub> has characteristics of high pressure and high refrigerating capacity. For CO<sub>2</sub> compressors with small displacement volume, the leakage loss in compression process increases caused by high pressure-difference. As a result, the compression efficiency declines.

We investigated the relationship between the performance and the oil-injection rate to the compression chambers, and found that the optimum oil-injection rate exists on every refrigerant flow rate. We also tried to develop the prototype of the CO<sub>2</sub>-scroll compressor with high compression efficiency, by adopting the high-pressure shell and using the optimum oil-injection rate to the compression chambers to decrease the leakage loss.

This paper presents the results of the experimental relationship between the performance and the oil-injection rate to the compression chambers. With optimizing the oil-injection rate, decreasing the leakage loss of refrigerant in compression process, our prototype of the CO<sub>2</sub>-scroll compressor achieved the higher efficiency of 102 -109% compared with the 2-stage rotary compressor that we developed.

## 1. INTRODUCTION

The production of fluorocarbons containing chlorines is to be phased out by the year 2020 to protect the ozone layer. The Kyoto Protocol was adopted to restrict the use of HFC refrigerants from the viewpoint of high global warming potential. On the other hand, movements have occurred to review the natural substances as refrigerants for heat pumps, and recently research and development of heat pump systems, using natural refrigerants is being actively promoted. In the circumstances, particularly carbon dioxide (CO<sub>2</sub>), with its advantageous non-flammable and non-toxic properties, is becoming a focus of attention from the viewpoint of low global warming potential, and aggressive works are being made to apply CO<sub>2</sub> to car air conditioners and hot water supply systems.

However, there are some problems, which must be overcome to improve the performance of the compressor efficiency and secure the reliability. CO<sub>2</sub> has characteristics of high pressure and high refrigerating capacity; the operation pressure is extremely high compared with R410A although the pressure ratio is relatively low. For CO<sub>2</sub> compressors with small displacement volume, the leakage loss in compression process increases caused by high pressure-difference. As a result, the compression efficiency declines.

We have developed the prototype of the CO<sub>2</sub>-scroll compressor. In the development process, we investigated the relationship between the performance of the compressor efficiency and the oil-injection rate to the compression chambers.

## 2. SELECTION OF THE COMPRESSOR TYPE FOR CO<sub>2</sub> REFRIGERANT

CO<sub>2</sub> refrigerant is a natural refrigerant, friendly to global environment, because the ozone layer destructive coefficient is 0 and the global warming coefficient is 1. Also from the non-flammable and non-toxic properties, it is attractive as an alternative refrigerant to take the place of fluorocarbons. On the other hand, its working

pressure is high and its high-low pressure difference is also large, the refrigerant density is high while the refrigerant volume flow rate is low. Selection of the compression type was made with consideration for such characteristics.

For the rotary compressors, it is possible to reduce the leakage loss if the clearances are set up small. But with structural limitations, it is difficult to set up the clearances much smaller comparing with the conventional rotary compressors. On the other hand, for the scroll compressors, by controlling the thrust force of the orbiting scroll, the clearance on the top of the wrap may be made small in operation and the leakage across the top of the wrap may be reduced. When the large force acts on the scroll wrap at the liquid compression or high load operating condition, the orbiting scroll will be pulled away from the fixed scroll to prevent the breakage of the wrap or abnormal wear.

Also, for the CO<sub>2</sub> compressor, the refrigerant density is high and the volume per capacity becomes about 1/3 times as much as that of R410A and therefore it is necessary to reduce the displacement volume. For the rotary compressors, there are two ways to reduce the displacement volume, reducing the height of the cylinder or reducing the eccentricity. There is the advantage of reducing the leakage loss on the cylinder side if the cylinder height is reduced but the effect of the vane groove distortion by high pressure becomes large. Also there is a problem that bearing load increases if the eccentricity is reduced. For the scroll compressors, reducing of the wrap height is conceivable. If the wrap height is reduced, the leakage of the wrap side may be reduced. Also, with increasing the rigidity of the wrap, the effect of heat expansion by compression heat may be reduced and therefore, there are no wear or increased shaft load by thrusting out of the wrap.

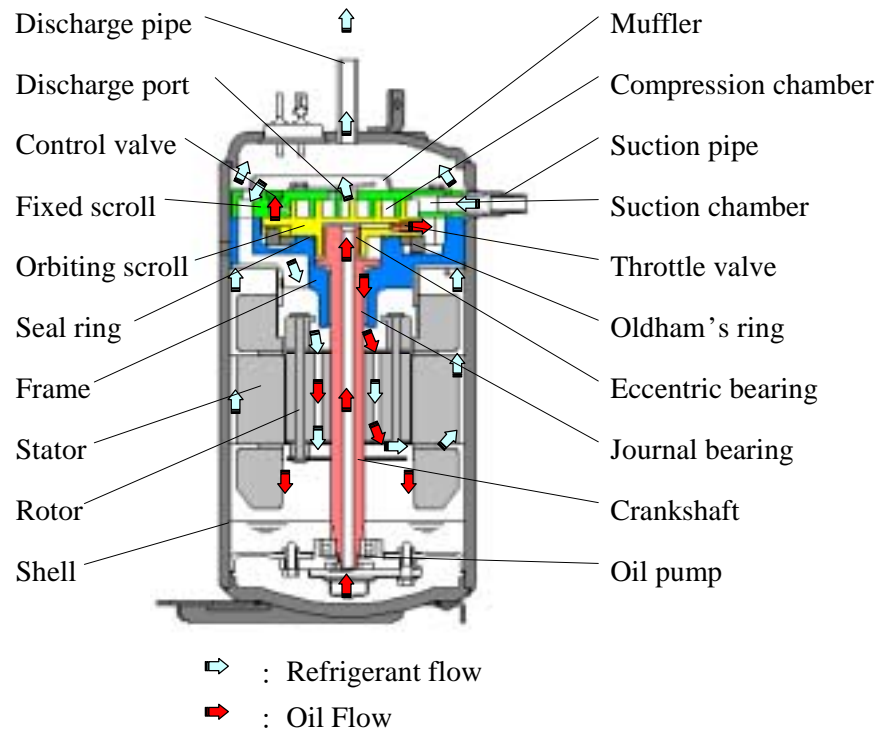
As mentioned above, for CO<sub>2</sub> refrigerant, it is possible that the scroll compressors may achieve the higher efficiency. Furthermore, from the principle of the compression mechanism, it possesses features such as lower noise and lower vibration. So, we selected the scroll compressors for CO<sub>2</sub> refrigerant.

**Table 1** Features of the compressor type

	Scroll type	Rotary type
High pressure difference	The clearance can be made smaller while securing the reliability.	It is difficult to make the clearances small with maintaining the reliability and performance.
Low displacement volume	The rigidity of the wrap may be increased while the leakage loss is reduced.	It is necessary to establish the structure to avoid the vane groove distortion or bearing over-load.
Other features	Low noise, Low vibration	

### 3. BASIC STRUCTURE OF THE PROTOTYPE

The present prototype was developed basing on our scroll compressor for R410A. We will explain the basic structure of the scroll compressor, using Fig.1.



**Fig. 1** Cross section of the scroll compressor

### 3-1. Explanation of the compressor structure

The fixed scroll part and the orbiting scroll part engage and form a multiple number of compression chambers. With the orbiting action, these compression chambers move towards the center while reducing their volume to compress the refrigerant. The orbiting scroll is connected to the crankshaft, which is driven by the motor. The Oldham's ring plays the role of preventing the self-rotation of the orbiting scroll. Also, the seal ring is located between the orbiting scroll and the frame to partition the high-pressure section (seal ring inner diameter side) and the intermediate pressure section (seal ring outer diameter side).

### 3-2. Refrigerant flow

The refrigerant returning from the refrigerant cycle passes through the suction pipe and is introduced into the suction chambers. The refrigerant confined in the suction chambers is compressed towards the center by the orbiting motion and discharged from the discharge port. The discharged refrigerant is introduced to the lower part by the muffler and led to the upper section via the connecting channel, and discharged from the discharge pipe and sent to the refrigerant cycle. During this process, the oil in the refrigerant is separated to prevent from being discharged into the refrigerating cycle.

### 3-3. Oil flow

The oil accumulating at the bottom of the shell is pumped up by the oil pump and passing through the inside of the shaft. The oil reaching the top of the crankshaft lubricates the eccentric bearings, main bearings in this order, and then returns to the bottom of the shell.

Also, some amount of the oil reaching the top of the crankshaft is depressurized by passing through the throttle valve installed in the orbiting scroll, and then reaches the periphery of the orbiting scroll. This oil acts to lubricate the Oldham's ring and to increase backpressure to prevent overturning of the orbiting scroll. The backpressure is maintained at the intermediate pressure by the backpressure adjusting mechanism installed in the fixed scroll. Then the oil flows into the suction chambers, and acts the sealing oil in compression process.

### 3-4. Specifications of the prototype

Table 2 shows the various specifications and Fig. 2 shows the exterior view of the CO<sub>2</sub>-scroll compressor with the above-mentioned basic structure.

**Table 2** Specifications

Compressor type	Scroll
Displacement	4cc
Refrigerant	CO <sub>2</sub>
Motor	DC Brushless



**Fig. 2** The exterior view of the CO<sub>2</sub>-Scroll compressor

## **4. EVALUATION OF CHARACTERISTICS**

### 4-1. Effect of the oil-injection to the compression chambers

If the oil-injection to the compression chambers is too low, the lubrication of the sliding part becomes insufficient and there is a risk of abnormal wear. Also the sealing effect in the compression chambers decreases and the leakage loss increases. As a result, the performance declines. On the contrary, if the oil-injection is too high, the suction heating of the refrigerant occurs and therefore the mass of the refrigerant (refrigerant flow rate) confined in the suction chambers decreases. Also in this case, the performance declines.

Using R410A as a refrigerant and by investigating the relationship between the oil-injection rate and the performance, attempts to optimize the oil-injection rate and improve the performance is reported [1] but when CO<sub>2</sub> is used as the refrigerant, the pressure becomes about 3 times and the volume per capacity becomes about 1/3 times as much as those of R410A. Therefore it is expected that the optimization value will change.

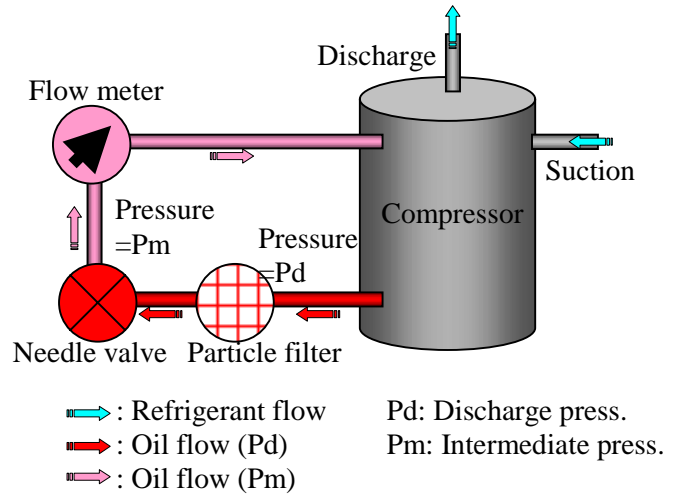
In this study, the relationship between the oil-injection rate and the performance of the efficiency with CO<sub>2</sub> refrigerant was investigated in detail and we tried to reduce the leakage loss of the scroll compressor.

### 4-2. Experimental apparatus

Fig. 3 shows the experimental apparatus to measure the relationship between oil-injection rate and the performance of the efficiency. The oil accumulating at the bottom of the shell is depressurized by the needle valve after passing through the particle filter. Then the oil flow rate was measured by the mass flow meter and supplied to the periphery of the orbiting scroll. At the same time, the throttle valve installed in the orbiting scroll is closed, so all of the oil passing through the mass flow meter passes through the backpressure adjusting mechanism, reaches the suction chambers, and becomes the sealing oil in the compression chambers. The needle valve is the variable needle valve and the oil-injection rate may be freely adjusted by changing the pressure loss. Also, the performance of the efficiency is measured by using the compressor evaluation apparatus.



Flow meter



**Fig.3** Experimental apparatus

The experimental results in this study were investigated under various operating conditions shown in Table 3.

**Table 3** Operating conditions

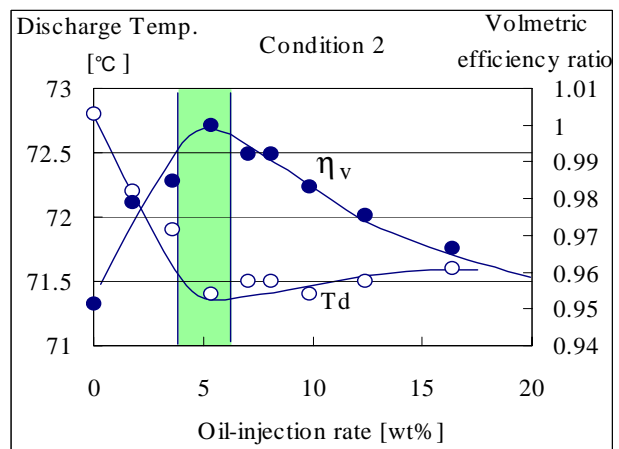
	High pressure [MPa]	Low pressure [MPa]	Compression ratio	Rotating speed [1/s]
Condition 1	8.0	3.8	2.1	17
Condition 2	9.0	5.0	1.8	37
Condition 3	10.0	4.0	2.5	62
Condition 4	9.0	3.0	3.0	62

## 5.RESULTS

### 5-1. Relationship between the oil-injection rate and the performance of the efficiency

Fig. 4 shows the experimental results of the relationship between the oil-injection rate and the volumetric efficiency ratio  $\eta_v$ , and the discharge temperature  $T_d$  at condition 2.

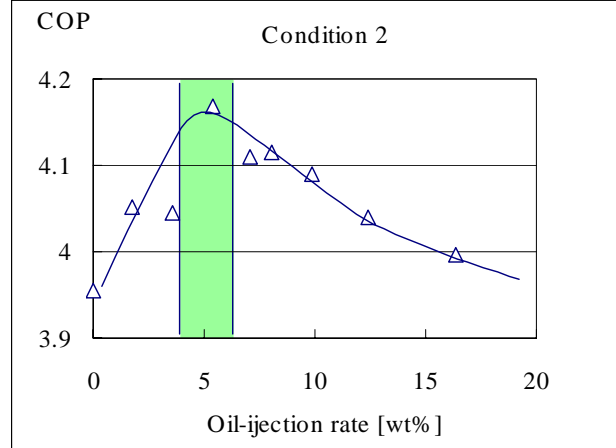
The oil-injection rate in the horizontal axis is obtained by dividing the refrigerant flow rate into the amount of the oil-injection to the compression chambers. For the volumetric efficiency ratio, it was found that there is a peak around the oil-injection rate of 5.4 wt %. It is thought that when the oil-injection rate is less than 5.4 wt %, the refrigerant flow rate decreases by the insufficiency of the sealing oil in suction process and by the leakage from the upstream compression chambers. It is also thought that when the oil-injection rate is more than 5.4 wt %, the refrigerant flow rate decreases by the suction heating. For that reason, the volumetric efficiency decreases. Also it was found that the discharge temperature increases greatly in proportion to the decrease of the oil-injection rate when the oil-injection rate is less than 5.4 wt %.



**Fig. 4** Relation between the oil-injection rate and the volumetric efficiency ratio, the discharge temp. at condition 2

That is the reason why that the leakage loss increases with the insufficiency of the sealing oil. Moreover, when the oil-injection rate is more than 5.4 wt %, the discharge temperature rises slightly. It is thought that the suction heating of the refrigerant in suction process occurs.

Fig. 5 shows the relationship between the oil-injection rate and the coefficient of performance, based on the above-mentioned measurement results. The coefficient of performance in the vertical axis is obtained by dividing the compressor input into the refrigerating capacity measured by the experimental apparatus. For the coefficient of performance, it was also found that there is a peak around the oil-injection rate of 5.4 wt % where the volumetric efficiency peaked and the existence of an optimum oil-injection rate was confirmed at condition 2.



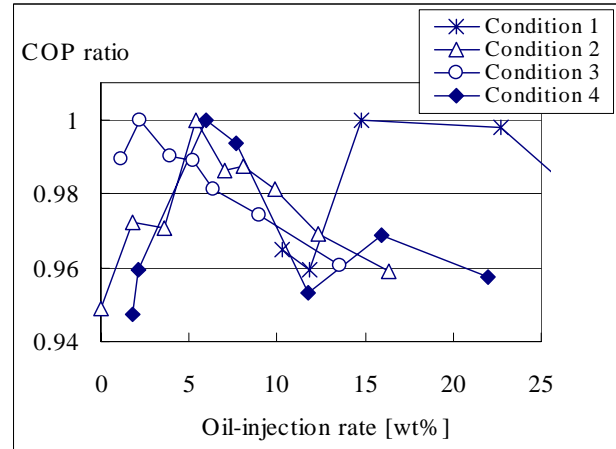
**Fig. 5** Relationship between the oil-injection rate and COP at condition 2

## 5-2. Characteristics of the optimum oil-injection rate

Similar experiments were conducted at the other three conditions. Fig. 6 shows the relationship between the oil-injection rate and the coefficient of performance at each condition. The coefficient of performance in the vertical axis is expressed in the coefficient of performance ratio obtained by dividing by the maximum value. In Fig. 6, the existence of the oil-injection rate where the coefficient of performance maximizes at each condition was confirmed. Table 4 shows the optimum oil-injection rate where the coefficient of performance maximizes at each operation condition.

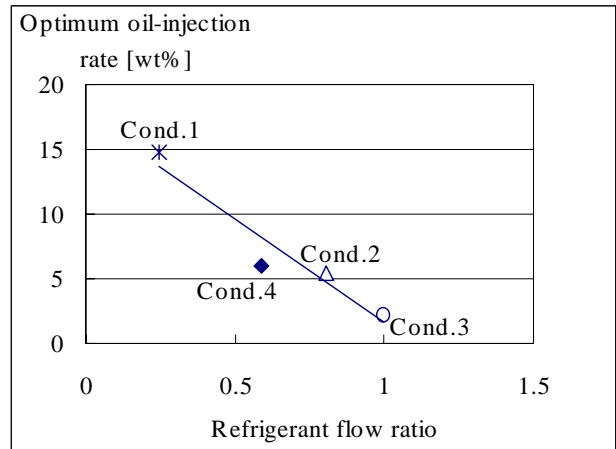
**Table 4** Optimum oil-injection rates

	Optimum oil-injection rates [wt%]
Condition 1	14.8
Condition 2	5.4
Condition 3	2.2
Condition 4	6



**Fig.6** Relationship between the oil-injection rate and COP at each condition

Fig. 7 shows the relationship between the refrigerant flow rate and the optimum oil-injection rate at each condition. The refrigerant flow ratio shown in the horizontal axis was obtained by dividing by the refrigerant flow rate of condition 3 because the refrigerant flow rate of condition 3 is the maximum value compared with those of the other conditions. In Fig. 7, it was found that the optimum oil-injection rate becomes larger as the refrigerant flow rate decreases and the optimum oil-injection rate becomes smaller as the refrigerant flow rate increases. It was found that although the various parameters relating to the leakage loss such as frequency, pressure conditions, compression ratio, discharge temperature, oil viscosity differ at each condition, the optimum oil-injection rate is strongly correlated to the refrigerant flow rate.

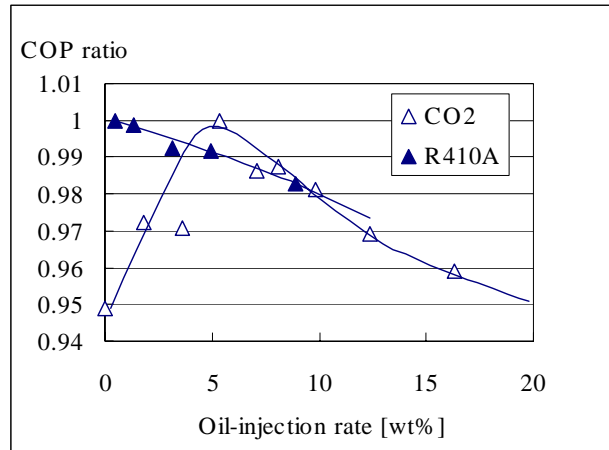


**Fig.7** Relationship between the optimum oil-injection rate and the refrigerant flow ratio

### 5-3. Comparison CO2 with R410A about the optimum oil-injection rate

For R410A, Sakuda et al., [1] investigated the relationship between the oil-injection rate and the performance of the efficiency in detail and achieved the performance improvement. Fig. 8 shows the relationship between the oil-injection rate and the coefficient of performance compared CO2 (Condition 2) with R410A. Here, the scroll compressor for R410A, designed so that refrigerant capacity and frequency would roughly approximate those of CO2 (Condition 2), was used to get the experimental results.

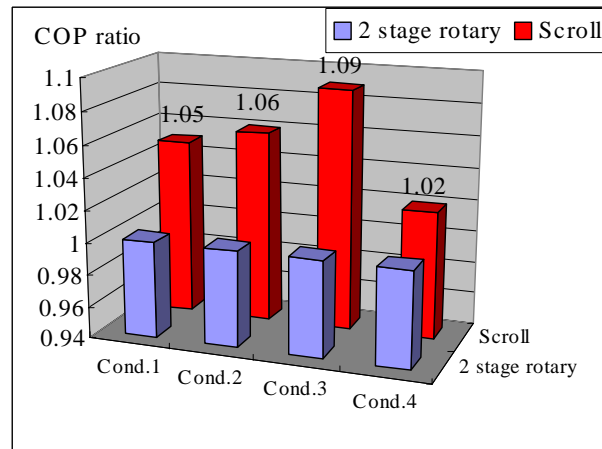
For R410A, no peak exists in the coefficient of performance and the efficiency increases as the oil-injection rate decreases. This tendency shows that the effect to increase the loss by the suction heating is predominant compared with the effect to reduce the leakage loss by injecting the sealing oil. On the other hand, for CO2, since the peak exists in the coefficient of performance, reducing the leakage loss by injecting of sealing oil is more effective and it was found that the appropriate oil-injection rate must be supplied to the compression chambers for the performance improvement.



**Fig.8** Relationship between the optimum oil-injection rate and COP (Comparison CO2 with R410A)

### 5-4. Comparison the scroll compressor with the 2-stage rotary compressor

For the prototype CO2-scroll compressor, besides the optimization of oil-injection rate to the compression chambers, the optimization of the element technology such as adjusting the backpressure, discharge port shape, relief valve position was done. As a result, it was found that compared with the 2-stage rotary compressor that we developed, the higher efficiency of 102 - 109% was achieved and it was confirmed that the scroll has the advantage for CO2 heat pumps.



**Fig. 9** Comparison the scroll compressor with the 2-stage rotary compressor about COP



## **6. CONCLUSIONS**

We developed the prototype of the CO<sub>2</sub>-scroll compressor. Also we investigated the relationship between the oil-injection rate to the compression chambers and the performance of the efficiency in detail. The results of the present study can be summarized as follows.

- [1] The optimum oil-injection rates at various operating conditions exist and the range of the optimum oil-injection rate (wt %) is 2 - 15.
- [2] The optimum oil-injection rate is strongly correlated to the refrigerant flow rate.
- [3] The scroll compressor achieved the higher efficiency of 102 - 109% compared with the 2-stage rotary compressor and it was confirmed that the scroll compressor has the advantage for CO<sub>2</sub> heat pumps

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